SDC SOLENOID DESIGN NOTE #140

TITLE: Investigation of Coil and Support Cylinder Interaction

AUTHOR: Ang Lee and Bob Wands

DATE: July 11, 1991

INTRODUCTION: This design note contains a study of the interaction between the coil conductor and support cylinder for the SDC solenoid detector. The objective of this study is to find the proper way to prevent the relative motion between the coil conductor and support cylinder when an axial magnetic force is exerted on the coil. Three cases have been studied. The first case considers a preload applied between coil and support cylinder through end contact. As the magnet is energized, a magnetic axial force will act on the coil and cause the coil to compress towards its center. This compression will allow the support cylinder to release its pretension and move back towards its initial position. The most interesting thing in this process is whether these two motions, coil compression and support cylinder tension release, will result in the same strain at a given point, meaning that the relative motion will be zero everywhere along the coil/support cylinder interface. The second case considered the use of epoxy glue to hold coil and support cylinder together. The maximum shear stress at the the coil/support cylinder interface is calculated and compared with the strength of the glue joint. The third case considers the use of frictional force between the coil and support cylinder to prevent the relative motion. A frictional force is calculated by using the normal pressure between the coil conductor and support cylinder and an assumed friction coefficient.

A geometry of the coil conductor and support cylinder is shown in Figure. 1

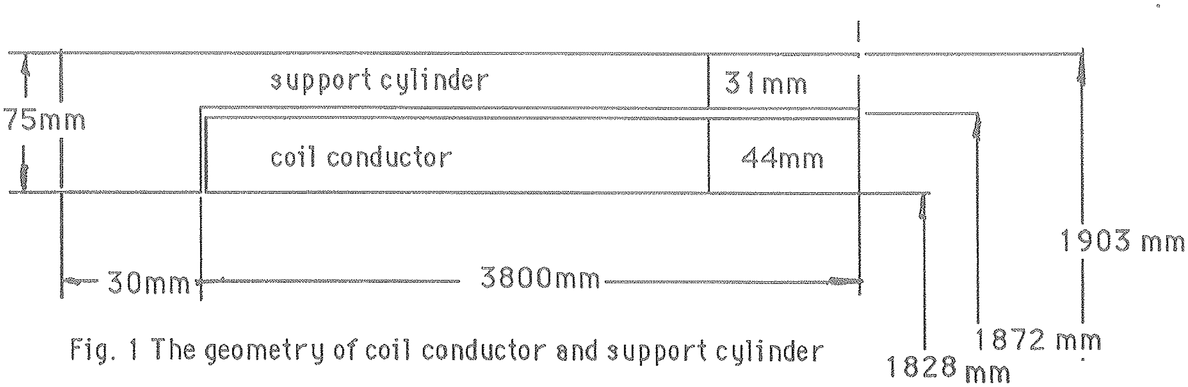

Fig. 1 The geometry of coil conductor and support cylinder

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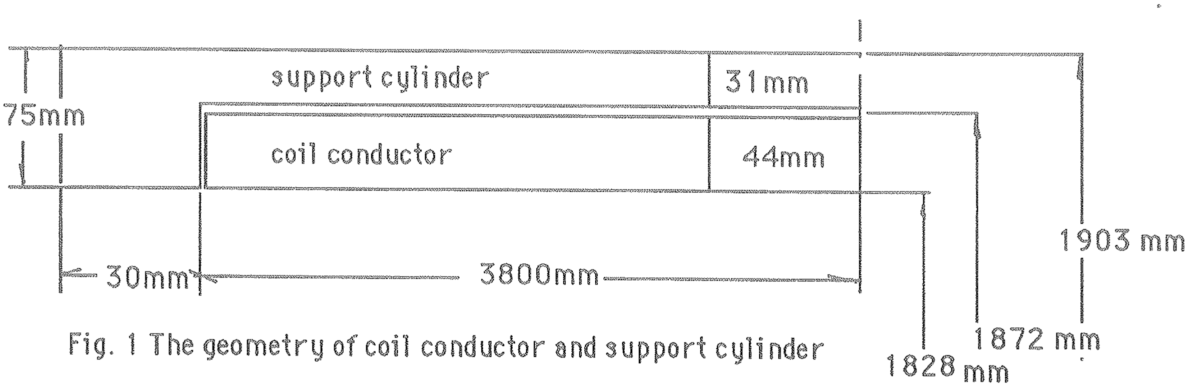


Fig. 1 The geometry of coil conductor and support cylinder

CASE STUDY

1. CASE 1 ____ Preload Both the Coil and Support Cylinder

As a first approximation, we assume that the preload causes the support cylinder to stretch and the coil conductor to compress as show in Fig. 2. To calculate the displacement of coil conductor due to the axial magnetic force, a infinitesimal element is taken from coil conductor with a length dx. The force acting on the cross section of this small element

$$F(x) = \frac{k_1}{k_1 + k_2} \int_0^x f(x)\,dx \qquad (1)$$

where the f(x) is a force density (N/m), k_1 and k_2 are the stiffness of coil conductor and support cylinder respectively

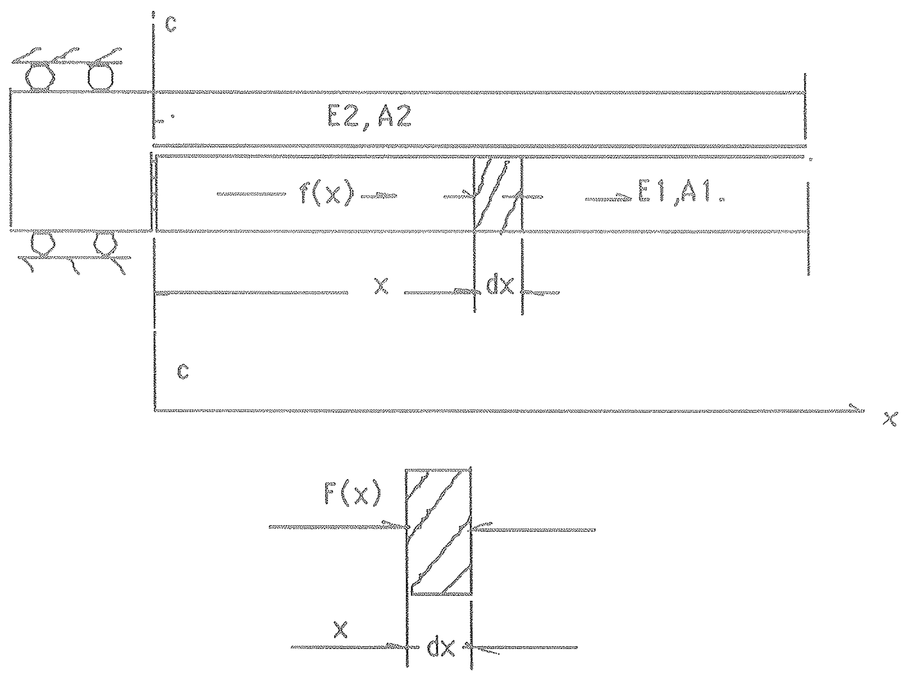


Fig.2 The model of the coil and support cylinder for "preload " case

The incremental stress due to the axial force will be

$$\sigma_1(x) = \frac{F(x)}{A_1} = \frac{\frac{k_1}{k_1+k_2}\int_0^x f(x)\cdot dx}{A_1} \tag{2}$$

and its strain

$$\varepsilon_1(x) = \frac{\sigma_1}{E_1} = \frac{\frac{k_1}{k_1+k_2}\int_0^x f(x)\,dx}{E_1 A_1} \tag{3}$$

The displacement of that small element is

$$d\delta_1(x) = \varepsilon_1(x)\,dx = \frac{\frac{k_1}{k_1+k_2}\int_0^x f(x)\,dx}{E_1 A_1}\,dx \tag{4}$$

The displacement of the coil conductor at position x will be

$$\delta_1(x) = \int_x^L d\delta_1 = \int_x^L \left\{ \frac{\frac{k_1}{k_1+k_2}\int_0^x f(x)\,dx}{E_1 A_1} \right\} dx \tag{5}$$

If the axial force is assumed to have exponential form, $f(x)=\alpha e^{-\beta}$,where α and β is a constant , the integration of equation 5 can be easily carried out as

$$\delta_1 = \left(\frac{k_1}{k_1+k_2}\right)\left(\frac{1}{E_1A_1}\right)\left(\frac{\alpha}{\beta}\right)\left((L - x) + \frac{e^{-\beta L} - e^{-\beta x}}{\beta}\right) \tag{6}$$

For x=L, equation 6 gives a zero displacement , which satisfies the boundary condition due to the symmetry at the center. For x=0, which is the end of the coil conductor, equation 6 gives a total displacement at the coil end due to the axial force. This end motion shrinks the conductor and allows the support cylinder to release its pretension. Since the case we are interested in is when these two ends maintain contact, the displacement of support cylinder δ_2 at C-C section will be equal to $\delta_1(x=0)$. The incremental force due to the end motion of the support cylinder will be

$$F_2 = \delta_2 \frac{E_2 \cdot A_2}{L} \tag{7}$$

The incremental stress will be

$$\sigma_2 = \frac{F_2}{A_2} = \delta_2 \frac{E_2}{L} \tag{8}$$

and corresponding strain due to axial magnetic force will be

$$\varepsilon_2 = \frac{\sigma_2}{E_2} = \frac{\delta_2}{L} = \frac{\delta_1(x=0)}{L} = \left(\frac{k_1}{k_1+k_2}\right)\left(\frac{1}{E_1A_1}\right)\left(\frac{\alpha}{\beta}\right)\left(L + \frac{e^{-\beta L} - 1}{\beta}\right)\frac{1}{L} \tag{9}$$

and its displacement

$$\delta_2 = \varepsilon_2 (L - x) = \left(\frac{k_1}{k_1+k_2}\right)\left(\frac{1}{E_1A_1}\right)\left(\frac{\alpha}{\beta}\right)\left(L + \frac{e^{-\beta L} - 1}{\beta}\right)\frac{L-x}{L} \tag{10}$$

By plotting equation 6 and equation 10 for a given material (see Figure 3), it shows that the relative motion (δ_1 - δ_2) is strongly affected by the axial force distribution f(x). The uniformly distributed axial force gives the largest relative motion. It is also noticed that

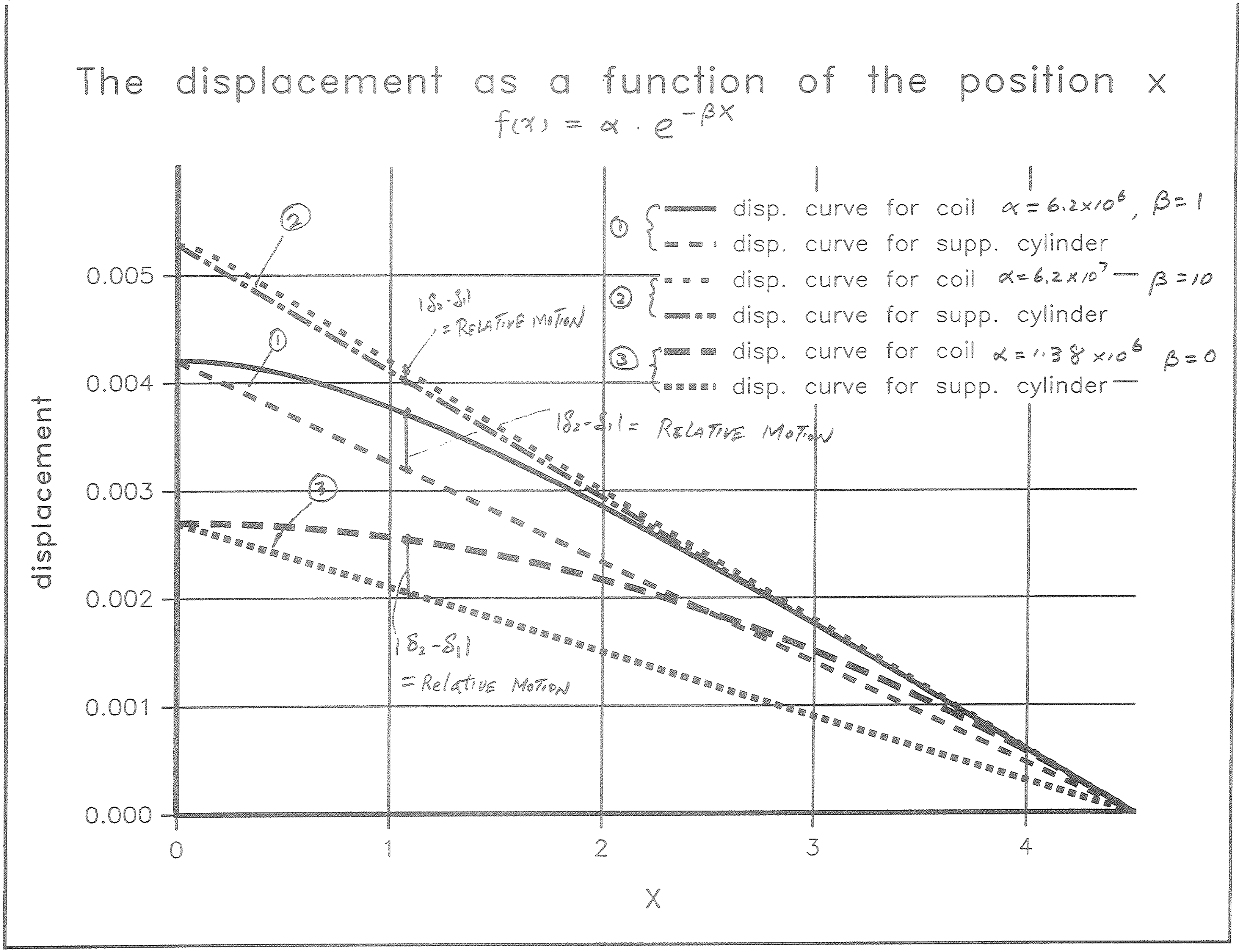


Fig.3 The sensitivity study of relative motion for different axial force distribution

these two displacement curves start to get closer as the axial force shifts down to the coil end. If we make β go to infinity, which represents a concentrated load acting on the end of coil, equations 6 and 8 become identical as

$$\lim_{\beta \to \infty} \delta_1 = \lim_{\beta \to \infty} \left(\frac{k_1}{k_1 + k_2}\right)\left(\frac{1}{E_1 A_1}\right)\left(\frac{\alpha}{\beta}\right)\left((L - x) + \frac{e^{-\beta L} - e^{-\beta x}}{\beta} \right)$$

$$= \left(\frac{k_1}{k_1 + k_2}\right)\left(\frac{1}{E_1 A_1}\right) F_0 (L - x) \tag{11}$$

and

$$\lim_{\beta \to \infty} \delta_2 = \lim_{\beta \to \infty} \left(\frac{k_1}{k_1 + k_2}\right)\left(\frac{1}{E_1 A_1}\right)\left(\frac{\alpha}{\beta}\right)\left(L + \frac{e^{-\beta L} - 1}{\beta} \right)\frac{L - x}{L}$$

$$= \left(\frac{k_1}{k_1 + k_2}\right)\left(\frac{1}{E_1 A_1}\right) F_0 (L - x) \tag{12}$$

where the Fo, as a total force, is defined as

$$F_0 = \int_0^L f(x)\,dx = \int_0^L \alpha \cdot e^{-\beta x} dx = \frac{\alpha}{\beta}(1 - e^{-\beta L}) \tag{13}$$

and

$$\lim_{\beta \to \infty} \frac{\alpha}{\beta} = \lim_{\beta \to \infty} \frac{F_0}{(1 - e^{-\beta L})} = F_0 \tag{14}$$

Therefore, the relative motion will vanish, and the two curves will collapse on each other as expected. The model will become two springs with different stiffnesses connected in parallel. However, the real axial force will be a distributed force, and its strain will be a function of x. The strain of support cylinder will, however, be a constant in the x direction because all action takes place in its end. Hence, these two displacement curves never be able to match each other. We conclude that the relative motion can not be prevented by "preload".

Case 2 ___Glue Joint Between Coil Conductor and Support Cylinder

A 2-D axisymmetric finite element model is constructed to calculate the shear stress in an epoxy bond between the coil and support cylinder. The distribution of the magnetic force on the coil is obtained directly from a finite element model magnetostatic calculation. The stress-stain curve for coil conductor is assumed to be that of the CDF coil with a strain limit 0.1% [2]. The shear strength of epoxy is assumed to be 2800 psi [3]. Five different geometries [4] of the coil have been investigated. The result are shown in Table 1. It shows that for the air _core case with an axial force 1530 (tonnes), the safety factor can be as high as 28. If the case_0 is prefered, the SF value can go up to 140.

Table 1 Calculation Result for Five Different Geometries

Case	F(axial) (tonnes)	Max. shear stress (psi)	Safety Factor ($2800/\tau_{max}$)	Max. Strain %
Air_core	1530	98	28	0.1000
Case_2	1064	87	32	0.0500
Case_3	583	75	37	0.0350
Case_7	272	36	77	0.0025
Case_0	140	20	140	0.0020

CASE 3 __Using Friction Force to Prevent Relative Motion

In the absence of preload or epoxy bond, we may use frictional force to prevent the relative motion of the coil conductor and support cylinder. The frictional shear stress at the contact surface must be at least equal to the shear stress calculated in case 2, which is shown in Appendix A for different cases . The friction force can be calculated by using the normal pressure between the coil conductor and support cylinder available from the finite element models, and a frictional coefficient which is assumed to be 0.2. The results for five different cases have been shown in appendix A. It shows that this frictional joint only works for the case of F(axial)=140 tonnes. Other cases show that the maximum friction stress is less than the maximum shear stress, especially in the end of the coil. It means that there will be a relative motion as the coil is energized.

CONCLUSIONS: It has been shown that preloading the coil against the support cylinder can not prevent relative motion between the coil and support cylinder due to the distribution of axial magnetic force. For the case of an epoxy joint, it is found that the safety factor on shear failure of the glue can be between 28 and 140 depending on the coil geometry. Finally, the study of a frictional joint with a friction coefficient 0.2 shows that this approach only works for the case with F(axial)=140 tonnes.

REFERENCE

(1) Bob Wands, "Finite Element Analysis of shear Stress Between Conductor and Support Cylinder Due to Magnetic Loads in An Air Core Solenoid", EAR-3, Dec. 1989, (unpublished)

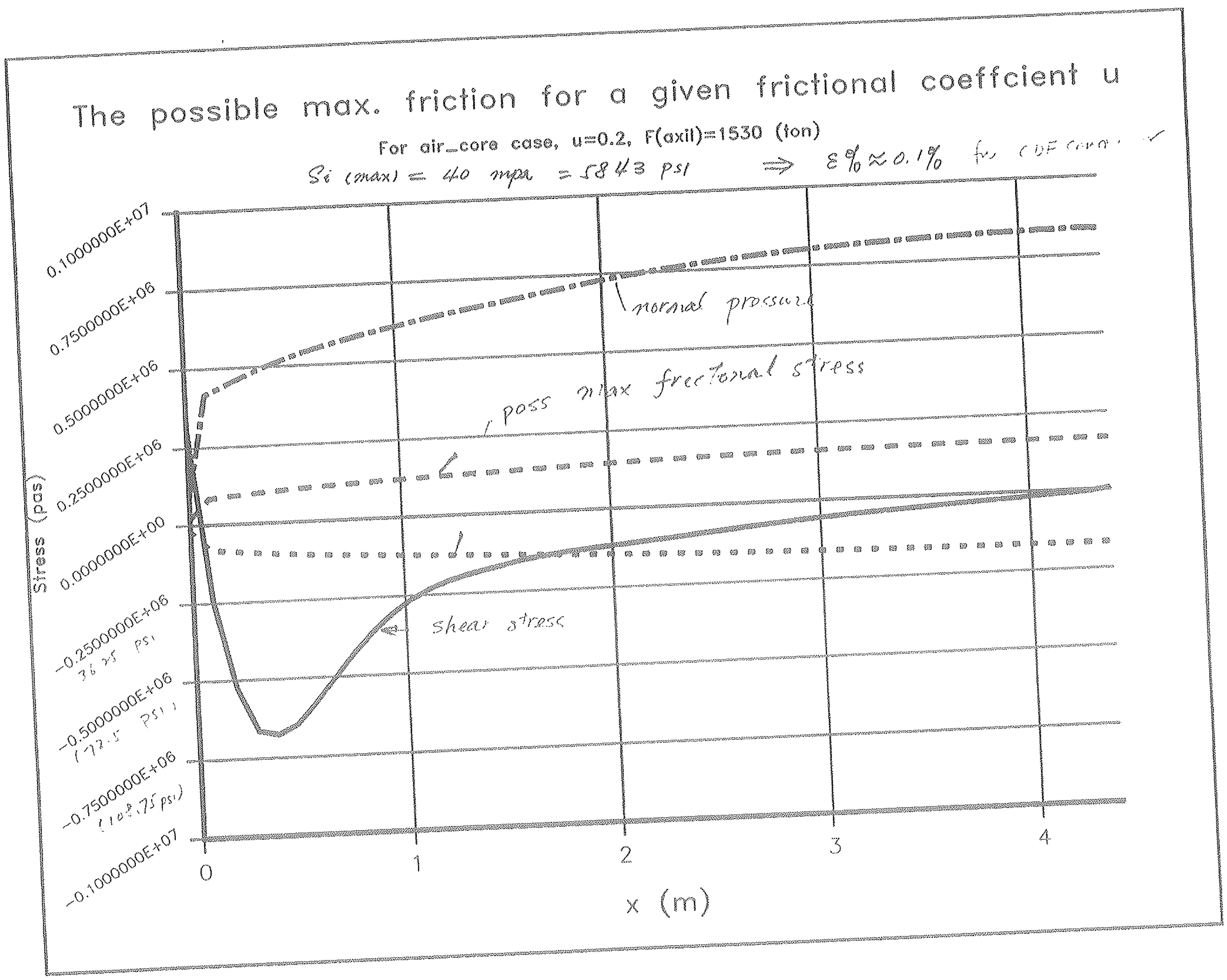
(2) "CDF Design Report", Fermilab, Oct. 1982 (unpublished)

(3) Ron Fast, etal , "Thin Solenoid Design Idea", SSC Detector Solenoid Design Note 101, Fermilab, Nov. 16,1989, (unpublished)

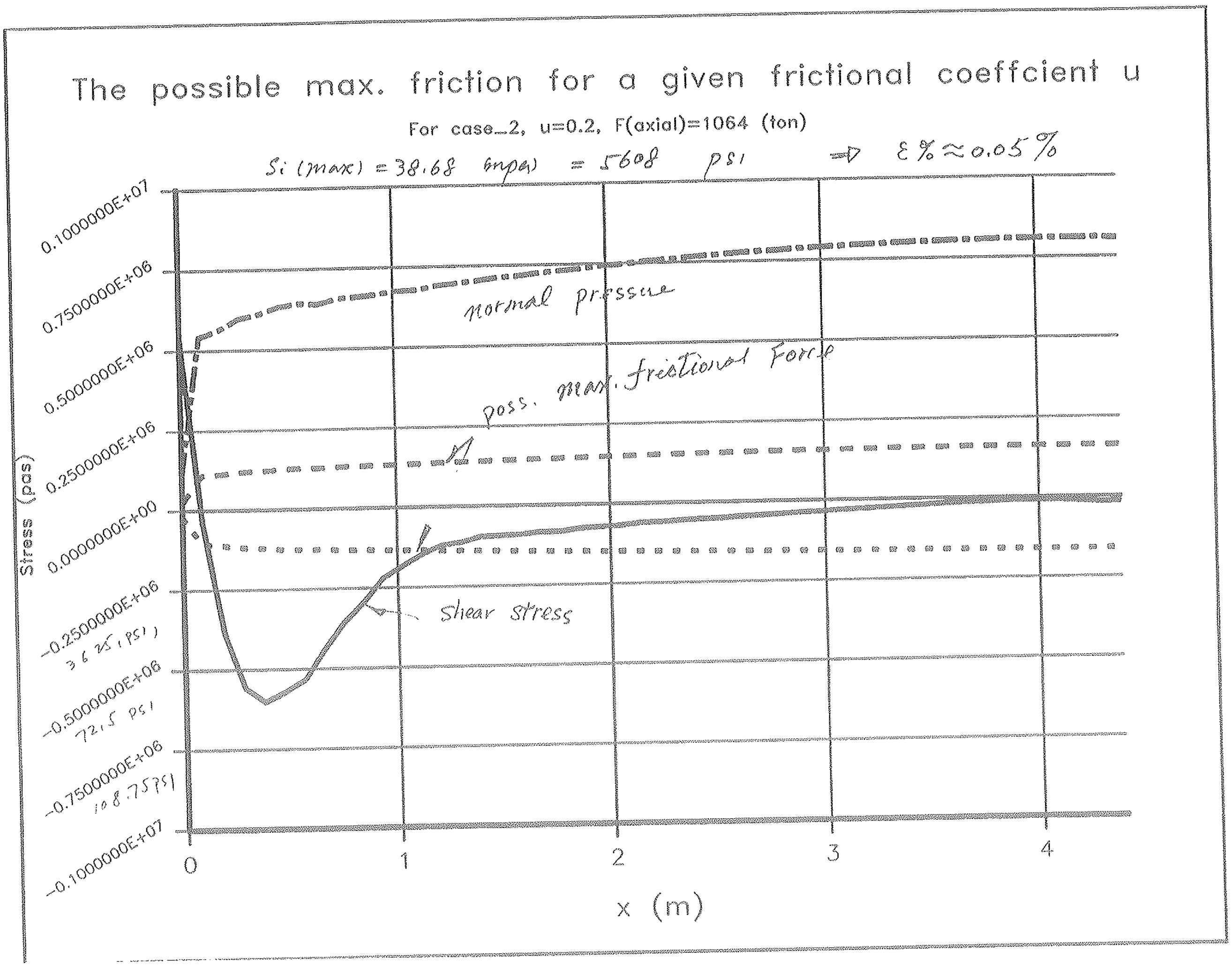
(4) Bob Wands, "Magnetostatic Analysis of Several SDC Solenoid/Calorimeter Configurations", SSC Detector Solenoid Design Note 138, Fermilab, March 22, 1991, (unpublished)

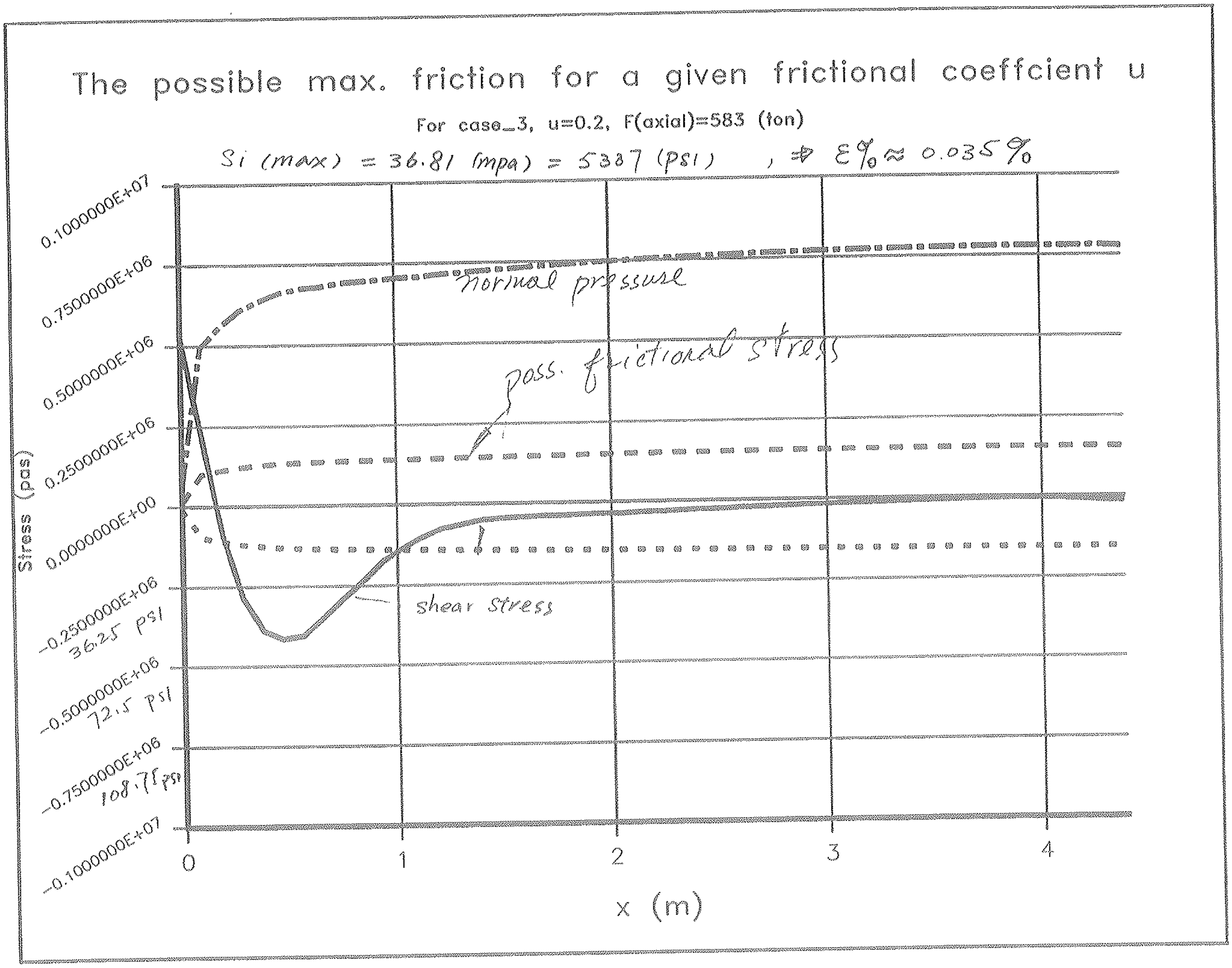
Appendices

a) The shear stress and maximum friction stress distribution for several different geometries

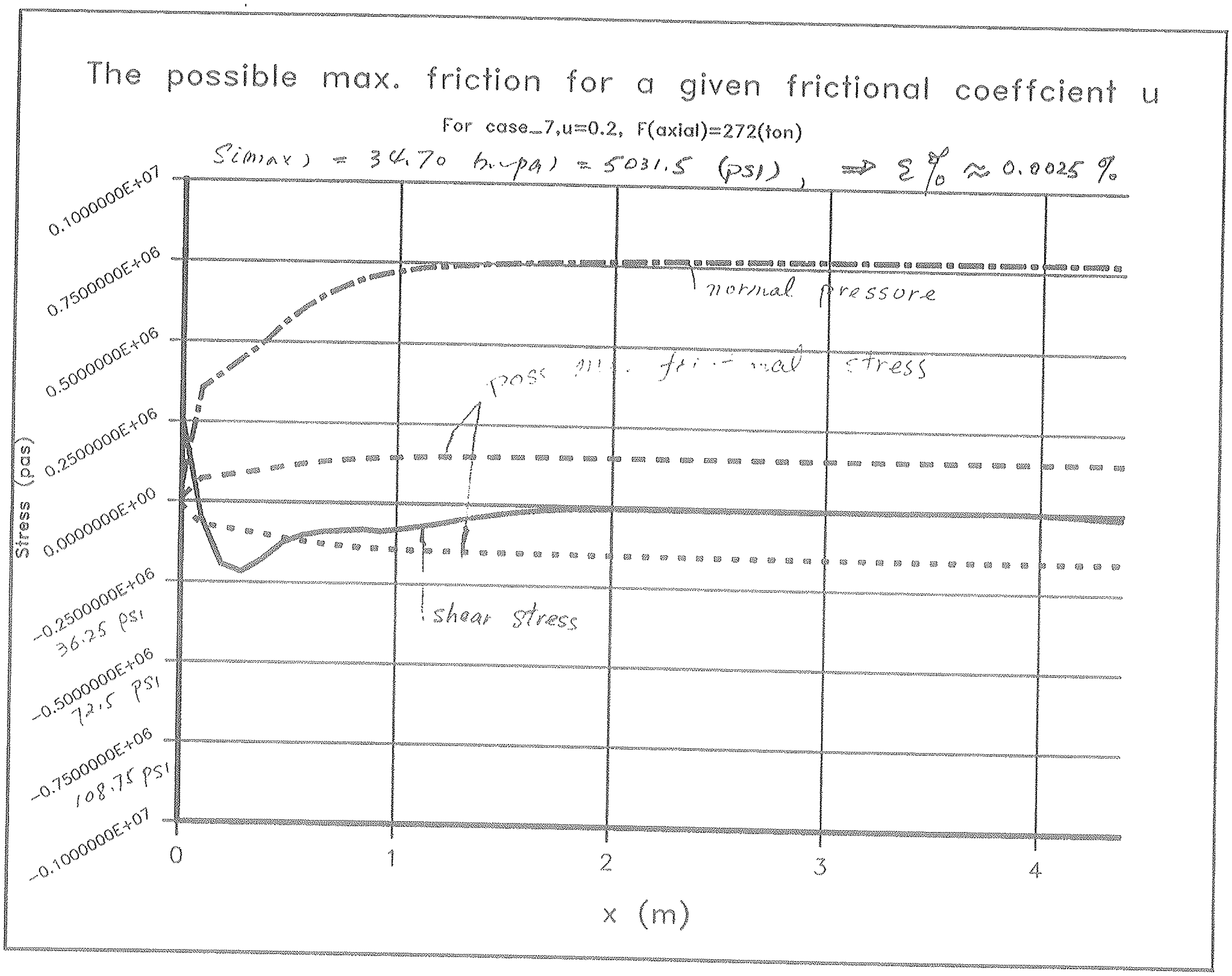


The possible max. friction for a given frictional coeffcient u
For air_core case, u=0.2, F(axil)=1530 (ton)
Si (max) = 40 mpa = 5843 psi ⇒ ε% ≈ 0.1%
normal pressure
poss max frictional stress
shear stress
Stress (pas)
0.1000000E+07
0.7500000E+06
0.5000000E+06
0.2500000E+06
0.0000000E+00
-0.2500000E+06
-0.5000000E+06
-0.7500000E+06
-0.1000000E+07
0
1
2
3
4
x (m)

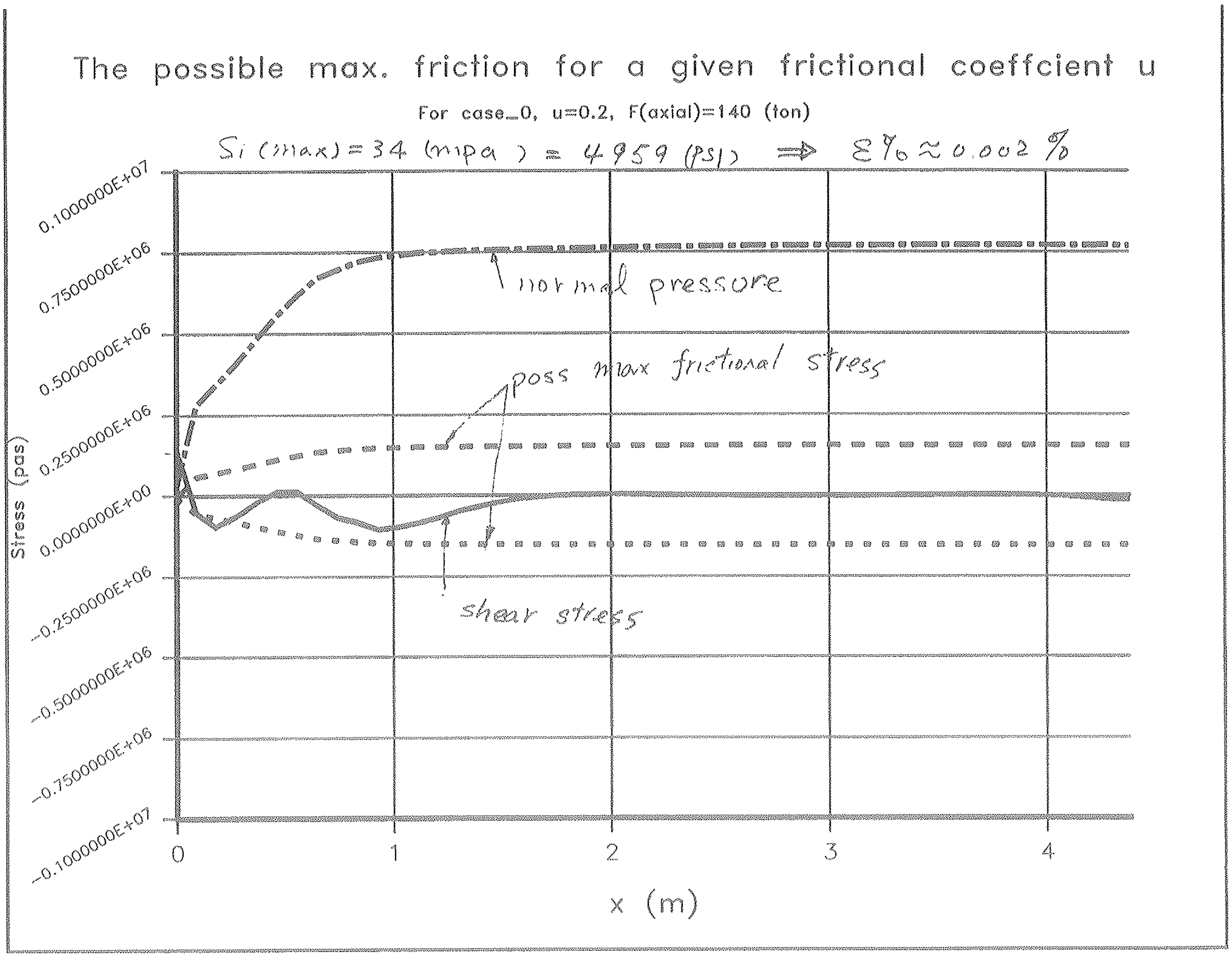

The possible max. friction for a given frictional coeffcient u
For case_2, u=0.2, F(axial)=1064 (ton)
Si (max) = 38.68 (mpa) = 5608 psi ⇒ ε% ≈ 0.05%
normal pressure
poss. max. frictional Force
shear stress
Stress (pas)
0.1000000E+07
0.7500000E+06
0.5000000E+06
0.2500000E+06
0.0000000E+00
-0.2500000E+06
(36.25 psi)
-0.5000000E+06
72.5 psi
-0.7500000E+06
108.75 psi
-0.1000000E+07
0
1
2
3
4
x (m)



The possible max. friction for a given frictional coeffcient u
For case_3, u=0.2, F(axial)=583 (ton)
Si (max) = 36.81 (mpa) = 5337 (psi) , ⇒ ε% ≈ 0.035%
normal pressure
poss. frictional stress
shear stress
Stress (pas)
0.1000000E+07
0.7500000E+06
0.5000000E+06
0.2500000E+06
0.0000000E+00
-0.2500000E+06
36.25 psi
-0.5000000E+06
72.5 psi
-0.7500000E+06
108.75 psi
-0.1000000E+07
0
1
2
3
4
x (m)



The possible max. friction for a given frictional coeffcient u
For case_7,u=0.2, F(axial)=272(ton)
S(max) = 34.70 (mpa) = 5031.5 (psi), ⇒ ε% ≈ 0.0025 %
0.1000000E+07
0.7500000E+06
0.5000000E+06
0.2500000E+06
0.0000000E+00
-0.2500000E+06
36.25 psi
-0.5000000E+06
72.5 psi
-0.7500000E+06
108.75 psi
-0.1000000E+07
Stress (pas)
normal pressure
shear stress
0
1
2
3
4
x (m)


The possible max. friction for a given frictional coeffcient u
For case_0, u=0.2, F(axial)=140 (ton)
Si (max) = 34 (mpa) = 4959 (psi) ⇒ ε% ≈ 0.002 %
normal pressure
poss max frictional stress
shear stress
Stress (pas)
0.1000000E+07
0.7500000E+06
0.5000000E+06
0.2500000E+06
0.0000000E+00
-0.2500000E+06
-0.5000000E+06
-0.7500000E+06
-0.1000000E+07
0
1
2
3
4
x (m)